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A REVIEW OF EVAPORATIVE COOLING SYSTEM CONCEPTS FOR ENGINE THERMAL MANAGEMENT IN MOTOR VEHICLES

by

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ABSTRACT

Evaporative cooling system concepts proposed over the past century for engine thermal management in automotive applications are examined and critically reviewed. The purpose of the review is to establish evident system shortcomings and to identify remaining research questions that need to be addressed to enable this important technology to be adopted by vehicle manufacturers. Initially, the benefits of evaporative cooling systems are restated in terms of improved engine efficiency, reduced CO₂ emissions, and improved fuel economy. An historical coverage follows of the proposed concepts dating back to 1918. Possible evaporative cooling concepts are then classified into four distinct classes and critically reviewed. This culminates in an assessment of the available evidence to establish the reasons why no system has yet made it to serial production. Then, by systematic examination of the critical areas in evaporative cooling systems for application to automotive engine cooling, remaining research challenges are identified.

1. INTRODUCTION

Evaporative Cooling (EC) is an effective means of thermal management and temperature control in a very diverse range of natural and man-made application areas - from the smallest of mammals to large electrical-power generation plants. Current engineering applications of this technology can be found in a number of important areas, for example: the air-conditioning of buildings in countries with low-humidity climates; in nuclear power plant reactors; and more recently, for cooling electronic hardware in personal computers. The application of EC to automotive combustion engine thermal management has seen, over the past century, numerous system concepts proposed, patented, analysed, prototyped, and in some cases implemented. Yet none of the proposed systems have actually made it to serial production. This review will examine the proposed concepts from an historical viewpoint and then put them into appropriately defined classes. It will then attempt to understand the reasons why (from a technology development viewpoint) existing EC system concepts have all effectively become stuck at a low technology readiness level with the result that none of the proposed systems have actually been implemented by vehicle manufacturers. The review will then address the remaining research challenges associated with EC systems, which if solved, will hopefully remove the major obstacles to implementation.

Figure 1 shows a schematic diagram of an automotive EC system. Water, in the liquid phase is introduced into the cooling jacket of the engine and a control system allows this water to boil off and become vapour. The liquid and vapour phases are separated in a separator tank and the vapour is returned to the liquid state in a condenser. A pump is used to circulate the flow around the system and to minimise pumping power, this pumps the liquid phase - not vapour.

The underlying principle behind all engine EC concepts is to exploit the substantial heat transfer rates that occur with the liquid-to-vapour phase-change that results during boiling. This provides a significant enhancement over conventional liquid-based engine cooling systems which remove heat largely through single phase convective heat transfer (although state-of-the-art 'subcooled' systems experience some breakthrough boiling). Engine cooling technologies, including conventional and subcooled systems have, from different perspectives, been excellently-reviewed in 2004 in [1], and in 2005 in [2], and more recently in 2010 [3]. Conventional systems are however reaching their limits for efficient thermal management, especially for aggressively-downsized highly-boosted engines. This limitation motivates the search for alternative cost-effective, efficient, durable, and controllable cooling systems. EC systems could overcome this limitation if the obstacles and challenges can be identified and overcome because, as discussed below, the benefits are clear.

It is clear that in an automotive engine EC system, the liquid-to-vapour phase change must take place within the cylinder jacket. In a typical concept design (and there are several distinct designs possible) the vapour formed must be vented, captured, and condensed from its gaseous state and returned to the cooling circuit as a liquid. Heat from the engine is therefore absorbed by exploiting the latent heat of vaporisation of the coolant. This results in several major advantages:

- 1) a reduction in coolant mass and overall system size;
- 2) lower coolant flow rates (with consequential lower pumping power losses);
- 3) uniform cylinder-head temperatures;
- 4) better knock control;
- 5) reduced noxious emissions;
- 6) reduced parasitic losses (e.g. cooling pump) and lower friction.

These advantages, in turn lead to improved overall engine efficiency, reduced CO₂ emissions, and improved fuel economy.

Over the past century (i.e. from the earliest evidence of EC system studies for automotive engine temperature control [4]), the period from 1918 to 1960 can be viewed as a modest period of pioneering work. During this time very few patents were granted, however, towards the end of the period (1958), a major US Navy report by Beck [5], comprehensively reviewed the prospects for EC of internal combustion engines. The period from 1962 to 1990 can be seen as one of more positive growth for automotive EC systems with five patents granted. During this period, i.e. in 1983, a comprehensive study of evaporative engine cooling was published by Leshner [6]. Since 1990 the level of growth has continued with nine patents being granted and the number of published research studies has also increased. For example, the significant analytical and experimental study published in 1997 by Porot et al [7] aimed to better understand and improve evaporative engine cooling at high engine loads and speed.

Although the fundamental physics of vapour-bubble formation within a saturated fluid is not yet fully understood [8] (especially the heat flux associated with bubble departure from a hot surface) it is still possible to logically classify EC systems concepts; the classification being based on system components. The main objectives of this review are therefore to help put the historical concepts into context by defining distinct EC system classes, and then to undertake a systematic examination of the critical areas in EC systems to identify the remaining research challenges for automotive engine cooling applications. To start off this process, a summary of heat transfer in boiling is given, followed by a brief discussion of simulation and modelling of boiling and two phase flow, culminating in a further technical discussion of the benefits of EC for IC engines.

2. THE PHYSICAL PRINCIPLES, CFD MODELLING AND BENEFITS OF EVAPORATIVE COOLING

2.1 Heat Transfer in Boiling

Boiling takes place at a solid-liquid boundary, distinguishing it from the process of evaporation which occurs at a liquid-vapour boundary. The physical mechanisms associated with the vigorous production of vapour bubbles and enhanced heat transfer rates take place when the heated surface temperature is higher than the saturation temperature of the liquid, T_{sat} . However, boiling also occurs when the bulk liquid temperature is below the saturation temperature. This is referred to as sub-cooled boiling - it is restricted to the thin layer adjacent to the heated surface, and as the vapour bubbles move through the sub-cooled liquid, they collapse and condense. Boiling without any externally-imposed flow or agitation is known as pool boiling. Flow boiling is the name given when there is superimposed flow or agitation. The heat transfer processes in an IC engine with evaporative cooling are expected to involve a combination of both pool and flow boiling. There are similarities between these two, thus it is appropriate to first consider the fundamentals of pool boiling before the more complex phenomena associated with flow boiling.

Figure 2 shows, for pool boiling of pure water at 1 bar, the heat flux q as a function of excess temperature $\Delta T = T_w - T_{\text{sat}}$, with both plotted on logarithmic scales [9]. The gradient of the curve gives the heat transfer coefficient. There are four distinct pool boiling regimes which occur as the excess temperature ΔT is increased, namely: i) free (or natural) convection, ii) nucleate boiling, iii) transition boiling, and iv) film boiling.

Free or natural convection, takes place with small excess temperature (typically $\Delta T < 4^\circ\text{C}$), where fluid motion is generated by buoyancy forces. Single-phase heat transfer correlations for free convection may be used in this regime. As ΔT is increased, individual

vapour bubbles begin to form at nucleation sites and rise through the body of the liquid. This marks the start of the nucleate boiling regime. A further increase in the excess temperature causes an increase in the production of these vapour bubbles, a corresponding increase in fluid motion, and a rise in the heat flux. The inflection point at $\Delta T \approx 10^\circ\text{C}$ is significant because it indicates a change from individual bubble formation to occurrence of large columns of vapour. The increase in thermal resistance associated with these larger entities causes a reduction in the rate of increase of the heat transfer coefficient. The heat flux reaches a maximum of $q_{\max} \approx 1.2 \text{ MW/m}^2$ at $\Delta T \approx 30^\circ\text{C}$. Beyond this, transition boiling occurs where an unstable film of vapour covers the surface. An increase in ΔT causes a reduction in the heat flux and a minimum value of heat flux, q_{\min} , occurs at the so-called 'Leidenfrost Temperature'. Beyond this, film boiling occurs with a stable film of vapour covering the surface. For these larger values of ΔT , radiation is significant and should be taken into account in heat transfer calculations.

Providing the relevant physical properties of the liquid and vapour phases are known, namely: dynamic viscosity, latent heat of vapourisation, surface tension, density, specific heat, and thermal conductivity, then the heat transfer rates in nucleate boiling and film boiling, and the values of the heat fluxes q_{\max} and q_{\min} can be obtained from the following well-established correlations i.e.: i) Nucleate boiling: Rohsenow [10]; ii) Film Boiling: Berenson [11]; iii) Maximum heat flux, q_{\max} : Lienhard and Dhir [12] and iv) Minimum heat flux, q_{\min} : Zuber [13]. These are also available in most heat transfer textbooks. In the nucleate boiling regime, the heat flux also depends on the nature of the surface, which is characterised by an empirical constant (usually tabulated).

For flow boiling, the different regimes of flow and heat transfer are generally delineated by the vapour quality (or dryness fraction) x . The heat flux in flow boiling is usually expressed as a summation of two contributions: i.e. single-phase forced convection, and

that due to boiling. For pure liquid ($x = 0\%$) entering a heated vertical tube, as the fluid proceeds upwards it will undergo nucleate boiling. Initially this occurs with the formation of individual bubbles, then with larger ‘slugs’ of vapour which eventually coalesce towards the centre of the tube forming an annular flow regime where a vapour core exists, the walls are coated with low thermal resistance liquid, leading to high heat transfer coefficient values which strongly depend on the fluid properties. For larger values of dryness fraction (typically for $x \approx 25\%$) there is a transition to a droplet flow regime. This is also associated with a significant reduction in the heat transfer coefficient as a consequence of the increase in the thermal resistance of the fluid (now vapour) adjacent to the walls. This occurs at the so-called “critical heat flux” and is of obvious significance to engine designers. Eventually (for a long enough tube), the vapour quality reaches 100%. Then, single-phase forced convection correlations may be used, based on the properties of the superheated vapour. A relatively simple correlation for flow boiling heat transfer in a vertical tube is given by Klimenko [14]. Qualitatively, the flow regimes inside a horizontal heated tube are similar to those in a vertical one. However, the interplay between the influence of buoyancy and fluid velocity serves to make delineation of the flow regimes more complex than in a vertical tube. Not surprisingly the physics of the flow behaviour and quantification of the heat transfer is more complex than in pool boiling. However Ghiaasiaan (2008) [15] provides an excellent review of flow boiling regimes and useful heat transfer correlations.

2.2 Simulation and Modelling of Boiling and Two Phase Flow

The boiling phenomenon and two-phase flow is a highly complex process. Multi-scale and multi-physical components are involved and interrelated, such as the nucleation, growth, departure, coalescence, and collapse of vapour bubbles, turbulence, interfacial instabilities, and heat transfer. Indeed, much of the physics is not yet fully understood. In

particular, it is not possible to mathematically describe the process of bubble nucleation in a deterministic way for flow boiling on real surfaces. It is therefore currently not possible to simulate boiling and two-phase flow directly. However, by making a number of appropriate approximations it is possible to undertake high fidelity simulation of boiling two-phase flow using the so called Direct Numerical Simulation (DNS) method [16-17] and the lattice Boltzmann (LB) method [18-19]. However, both methods are computationally excessive and applying them to boiling two-phase flow for real surfaces is currently impossible.

Some researchers favour modelling boiling flow as a statistical process [20-21] rather than using the deterministic approach of mechanistic modelling. The stochastic approach describes the uncertain fluctuations of the surface temperature associated with the non-linear interaction of bubble nucleation on neighbouring sites. However, a full predictive wall heat flux model has not yet been developed. Consequently the deterministic approach remains the only viable option.

The current state-of-the-art CFD methodology for the prediction of boiling two-phase flow involves computing the flow using the so called Reynolds-Averaged-Navier-Stokes (RANS) approach combined with a variety of wall heat flux models developed for use in engineering applications [22]. Broadly speaking the CFD models for the prediction of boiling two-phase flow can be classified into the following three categories:

- i) Incompressible single phase flow models: the flow is treated as a single phase with a modified thermal boundary condition to account for the heat transfer enhancement as a result of boiling, with empirical correlations for the wall-temperature/heat-flux under the boiling condition. There are some major drawbacks of this single phase approach such as the energy addition that will translate directly into a rise in temperature rather than phase change, leading to inaccurate

predictions of density, temperature, and the flow field. Therefore this method has limited application.

- ii) Homogenous flow models (also called a homogeneous mixture models) proceed under the assumption that vapour bubbles are small and are perfectly mixed with the liquid phase. A homogenous flow model can be used to represent both the liquid and vapour phases. The modelling equations describing mass, momentum, and energy conservation of the mixture, have the same form as the single phase equations with an additional variable called the 'void fraction' or 'volume fraction' being introduced to describe the concentration of the vapour phase. This method takes full account of the effect due to the fluid phase change but the detailed interfacial dynamics between the two phases is not properly modelled. Shala [23] used this method in conjunction with a mechanistic wall heat flux model to study nucleate boiling flow in a horizontal channel, and in a vertical annuls. The predictions are in broad agreement with the measured data. Li et al [24] applied a homogeneous flow model coupled with an empirical correlation for the wall heat flux to study boiling heat transfer in an engine cooling passage.
- iii) Eulerian two-fluid models include two sets of governing equations for the liquid and vapour phases, which are solved with the mass, momentum and energy transfer between the two phases being explicitly modelled. However, if the size of vapour bubbles is the same or smaller than the mesh size, the phase boundary cannot be predicted and hence the interactions between phases are approximated based on the locally estimated bubble size and number. When the bubble size becomes larger than the mesh size, details of the phase boundary can be predicted with the help of an interface treatment. Tu and Yeoh [25] undertook a CFD study of subcooled boiling flows using an Eulerian two-fluid model. The wall heat flux was calculated based on

an empirical correlation model. The agreement between prediction and experimental data was generally good. Narumanchi et al [26] applied an Eulerian multiphase model in combination with a mechanistic wall heat flux model to study nucleate boiling in impinging jets. Reasonable agreement between the experimental boiling curves and those obtained by CFD was obtained.

Another very popular model for two-phase flow is called the Volume of Fluid (VOF), which solves a single set of momentum equations and tracks the volume fraction for each of the fluids in each computational cell. The VOF is based on the fact that two fluids (or phases) are not interpenetrating. Therefore it is not used in boiling two-phase flow calculations.

All the CFD models mentioned above need input from a wall heat flux model to compute the wall heat flux. These wall heat flux models can be broadly grouped into two categories:

- **General empirical correlations**, which obtain the wall heat transfer rates as general power functions of a set of non-dimensional groups [27-31]. This kind of model relies completely on experimental data to derive the non-dimensional model correlations for wall heat flux based on curve fitting the experimental datasets, It is therefore only capable of predicting the total wall heat flux. Applications are usually limited, but may work well in certain cases. However since they do not account for the physical mechanisms involved, significant differences between prediction and experimental measurement can occur when conditions for which they were developed are not satisfied.
- **Mechanistic models**, which attempt to capture the total heat flux based on the individual heat transfer mechanisms involved, i.e. the hydrodynamic convective transport, and the thermal heat transport associated with evaporation [32-35]. This kind of model accounts explicitly for the different physical mechanisms contributing to the total heat flux, leading to better performance in general. However these models

still involve a good deal of empiricism in the sub-models for individual mechanisms. A comprehensive review on both empirical correlations and mechanistic models has been given by Warrier and Dhir [36].

2.3 The Benefits of Evaporative Cooling for IC Engines

The time-averaged cylinder head heat flux in down-sized spark ignition (SI) engines, rated at 100 kW/L, has already reached 1.25 MW/m². As shown in Figure 2 even for highly pressurised coolant systems, this rate of heat flow is close to the maximum heat flux associated with pool boiling. As EU urban drive cycle carbon dioxide limits are reduced to below 95 g/km, down-size ratings will increase and a combination of these higher ratings and the trend to lower engine speeds will lead to increased operational times at these high levels of heat flux. This engenders the following fundamental design challenges for light duty engines. The cylinder head gas side metal temperatures will rise beyond the thermal fatigue capability of traditional aluminium casting alloys. Steam venting and condensing rates will exceed the limits of single phase convection flow based coolant jackets and traditional “de-gas” systems. The stability and longevity of mixed aqueous/alcohol based coolants will be severely challenged by the expected high frequency changes of phase. Finally, the incidence of cavitation damage to static and moving parts and air ingress into the cooling system will increase with prolonged boiling. These issues justify rephrasing the question originally put in 1969 for heavy-duty compression-ignition (CI) engines [37] namely: “how will the heat be taken out of highly boosted spark-ignition engines?”

In addressing this question, there are at least two main routes to reduce the heat flow into the cylinders whilst development trends lead to increasing the fuel flow rate. Firstly, charge dilution is very effective in reducing heat losses from the cylinder. In this context, it is generally accepted that turbocharged compression ignition (CI) engines are naturally better suited to high boost than their SI counterparts. This is in part because auto-ignition

is the CI source of combustion, partly because high air/fuel operation with reduced combustion temperature is entirely natural and feasible with CI, and because reducing heat flow from the working fluid to the cylinder walls is easier with CI.

Boosted homogenous-charged SI engines are to some extent, successfully mimicking boosted CI engines. This is achieved by extensive air intercooling, two-stage turbocharging, and the use of dilute mixtures at stoichiometry with large levels of cooled exhaust gas re-circulation [38]. The latter is employed to avoid NO_x after-treatment by significantly reducing combustion temperatures, which not only reduces NO_x inside the cylinder but also mitigates knock.

Whilst these in-cylinder measures will significantly reduce thermal loading in SI engines, and ease localised coolant boiling challenges, the progressive increase in downsized SI ratings to 150 kW/L is expected to lead back to excessive values of local heat flux. This will make it necessary to consider alternatives to state-of-the-art mixed single phase convection/nucleate boiling cooling systems. The second route forward then, in addressing thermal loading, is to consider alternatives to mixed single phase convection / nucleate boiling systems.

Amongst potential solutions and palliatives to these thermal challenges, EC systems, as described in the following pages, warrants serious consideration as it offers major functional advantages. For instance, EC can provide controlled local boiling at a setpoint temperature across all the metal surfaces. This leads to more uniform metal temperatures and reduced spatial thermal gradients which in turn help to reduce low cycle thermal fatigue. Another advantage is that evaporation and condensation, with appropriate control, have significantly higher heat transfer coefficients than single phase convection cooling. Further benefits include the possibility of much reduced coolant quantities, very large reductions in coolant flow rates and parasitic pumping losses, improvements in knock

limited performance and reduced warm-up times with reduced pollutants and CO₂. Not surprisingly, there are some major challenges in implementing EC systems for automotive SI engines.

This paper will consider different types of existing, and possible future EC system concepts in Section 4 by classifying them according to an appropriate description of their key concept features. Before that, previous system solutions are reviewed from an historical perspective. The question is also addressed which, against a background of being used widely elsewhere, asks why EC systems are not currently being used for automotive IC engine thermal management.

3. AN HISTORICAL REVIEW OF AUTOMOTIVE EVAPORATIVE COOLING CONCEPTS

The fluid used in an EC system can either be water, or some mixture formulated to provide a desired boiling (and freezing) temperature. The EC mechanism, which makes use of the latent heat, allows a vapour-cooled engine to run at relatively uniform temperatures regardless of operating conditions. It was precisely this understanding that led the pioneers of EC systems to explore its possibilities. The historical progress in the development of EC systems for automotive applications can for convenience be divided into three phases:

Phase one: pioneering work. Between 1918 and 1960, the first theoretical and experimental studies were undertaken resulting in some landmark applications of EC systems. The internal combustion engine itself underwent very significant development throughout this period, therefore not surprisingly, the published findings concentrated on the fundamentals of the EC technology, and the possibility of application to production vehicles.

Phase two: growth. Between 1960 and 1990, comprehensive studies based on detailed testing and real applications were published. Volkswagen, Nissan, and Ford all filed international patents during this period.

Phase three: integration. From 1990 to the present day, more detailed experimental results were published which largely focused on modification of previous concepts, involving system integration and optimisation to achieve CO₂ reduction, better fuel economy, and reduced harmful emissions.

Table 1 shows key EC system milestones that were reached during these three phases - the most important publications during Phases 1 - 3 are then in discussed in detail.

Table 1. Important milestones in evaporative cooling system concepts

Year	Milestone
1918	Harrison first learned about evaporative cooling system from Muir who patented in 1922 [39].
1926	Harrison's SAE paper 'Evaporative Cooling' advocated use in passenger car [4].
1958	E. J. Beck's U.S. Navy report on Evaporative Cooling of Internal Combustion Engines [5].
1958	Dow Chemical's new coolant (Dowtherm 209) for evaporative cooling (See [5]).
1983	Leshner's SAE paper on evaporative cooling – survey and fuel economy consideration [6].
1986	One of Many Nissan patents for automotive evaporative cooling [48].
1987	Nissan Motor Concept truck with evaporative cooling system [52].
1993	Volkswagen patent for ICE evaporative cooling system [59].
2004	Pang and Brace comprehensively review ICE cooling systems [1].
2014	General Electric patent for integrated cooling system and method for engine powered unit [68].

Phase One: Pioneering Work.

The precise details are uncertain but it would appear that the first reported implementation of an EC system occurred around 1918 by W. W. Muir (resulting in the grant of a US patent in 1922) [39]. The basic concept was uncomplicated, but many practical considerations had to be taken into account for effective operation. The first experimental

test of an EC system took place in 1919 on a Packard car, which was driven for more than 2000 miles (through ice and snow) and reported to operate very satisfactorily. The first actual attempted use of EC in a production vehicle is reported by Harrison in 1926 [4]. Harrison claimed that EC systems could overcome all the shortcomings of conventional cooling, which included: the tendency of the system to become stagnant (presumably when left idle); the excessiveness of the radiator size; and the problem of crankcase oil dilution resulting from overcooling. It had in fact been recognised as early as 1919 that to realise the benefits of an EC system, important design considerations had to be taken into account such as the need for rapid circulation of the coolant, provision of an air lock, and the need for an 'efficient' condenser. These early systems used 'hopper' cooling which was the most elementary form of EC. An example of the concept is shown in Figure 3, in which the engine cylinder block is completely surrounded by a bath of water. The early systems were followed by use of a steam condenser as shown in Figure 4, fitted to the cooling hopper as a modification to improve cooling efficiency and to eliminate the need for regular topping-up of cooling water. The subsequent adoption of a thermostat and a pressure cap, successfully converted an otherwise open system into a closed cooling system (which in fact became the standard up to the present day, for all conventional liquid based engine cooling systems). During the 1920s and 30s, there were a number of patent applications for EC systems particularly for stationary power plant and marine engines, but apart from the patents being granted in 1931 [40] and 1947 [41], very little has been published reporting the success (or failure) of these developments. However in 1958, E. J. Beck [42, 5] completed a major study of EC technology for the U.S. Navy, providing a very thorough and detailed assessment of its potential for use with internal combustion engines, indeed laying the foundation for the next generation of developments. This was achieved by careful examination of the physical principles concerned with heat transfer through

boiling and condensation, the key design issues, and an assessment of state-of-the-art coolants at that time.

Phase Two: Growth

In the early 1960s, a patent filed by Yamaoka in 1962 [43] succeeded in renewing interest in EC systems. The following year, Bullard [44] patented a vapour-phase cooling system for an internal combustion engine. At the same time, the US Dow Chemical Company became interested in EC. They developed a new coolant called 'Dowtherm 209' (see [5]) which was formulated specifically for EC. It had the particular property of forming an azeotropic with water, meaning that water is not distilled out of solution when a mixture of water and Dowtherm 209 is used as the coolant.

Work reported in 1964 [45] confirmed the suitability of the essential features of the concept proposed by Harrison [4], but identified the main drawback of Harrison's design as being the lack of arrangements being made to cool the engine lubricating oil. The idea of dual circuit cooling was presented to involve an engine and condenser circuit, each being responsible for a specific working condition. The idea of a dual circuit would appear to successfully replenish the engine cooling jacket with water from two sources: water separated from wet steam, and water obtained by condensing steam. This proposed dual circuit system (shown in Figure 5) was installed and tested in a 6-110 GM diesel engine, with the results exhibiting the following benefits: uniform metal temperature, uniform coolant temperature, and improved fuel economy.

Between 1967 and 1987, a total of four US patents were granted [46-49] for new EC system concepts. In the 1983 US patent granted to Evans et al [47], they developed an EC system which operated at virtually constant (predetermined) pressure and temperature, using a condenser where the vaporised coolant was condensed under all operating conditions. Three years later Kubozuka [48] patented a similar concept that was claimed to

reduce the size of an engine through a reduction in the liquid coolant reservoir, and in the size of the separator. Yet another EC system concept patent appeared in 1987 [49] produced by Nissan focusing on an improved system that was claimed to solve the drawbacks of previous designs. This used a circuit for the coolant, and a variable capacity coolant tank, connected to a lower tank to capture the liquid coolant.

Earlier in 1983, Leshner [6] had fully reviewed the history of evaporative engine cooling methodologies and addressed its potential impact on fuel economy. He also studied the consequences of enhanced rates of heat transfer, such as uniformity of temperature, engine warm up, corrosion, combustion chamber deposits, and friction. By addressing the main drawbacks of [45] (in particular the relatively large system volume) Leshner developed an integrated separator tank and condenser system. This was given the name: 'Vacor Engine Temperature Management' or VETM, which was evaluated using several standard test procedures. The VETM system was tested on a 1980 Oldsmobile Omega, also a 1981 Volkswagen Rabbit Diesel, demonstrating improvements in both fuel economy and emissions. Experimental results were discussed in detail and the conclusions reached were:

- i) the benefits of EC are faster warm-up, and reduced friction resulting in better fuel economy, and elevated engine durability,
- ii) engine coolant temperature could be raised without increasing peak metal temperatures owing to a highly uniform temperature throughout the engine block,
- iii) engine parts are maintained above the dew point temperature after warm up which also reduces the period of dangerously low engine temperatures during warming up,
- iv) a reduction in the accumulation of combustion chamber deposits, including those in the lubricating oil,
- v) a higher oil temperature and lower viscosity, and consequently, a reduction in friction and improved fuel economy. This effect is enhanced by the rapid warming up characteristics of EC.

In 1987 Ford UK, in collaboration with the National Engineering Laboratory, published an experimental study on nucleate and film boiling of engine coolants [50]. A schematic diagram of the test rig used is shown in Figure 6. The work focused on improving understanding of the conditions that lead to the onset of film boiling. As shown in Figure 2, an increase in the surface-to-fluid-temperature difference ΔT , causes a reduction in heat flux, which then further increases ΔT . Prolonged operation in this regime could result in damage to the cylinder head. Measurements of heat transfer for water/ethylene glycol mixtures were reported for coolant velocities from 0.1 m/s to 5.5 m/s. At high coolant velocities, forced convection was the dominant mode of heat transfer. At lower velocities, strong nucleate boiling occurred. At the lowest velocity, 'dryout' or 'vapour blanketing' of the test section was detected. The main findings were that near the onset of film boiling: i) very unsteady coolant flowrates occurred resulting in great difficulty in maintaining a constant pressure-drop across the test section, and ii) as a result of an increase in heat flux, the increase in inner-surface temperature became highly erratic (at 'dryout' occurring continuously).

A Nissan sponsored study on EC systems is reported in [51-52]. The 1987 SAE publication [52] is rich with experimental data and analysis on different aspects of EC systems and its effect on internal combustion engine performance. For example, from a cylindrical pool boiling type rig, both the temperature gradient within the cylindrical portion, and the vapour temperature, were measured and correlated with calculated wall surface temperature values and gradients, heat transfer coefficients, and the heat flux levels. Observations were also made and reported of the inside of a condenser tube using the rig shown in Figure 7. The results were then applied to design an EC system for a 1.8 L four cylinder gasoline engine. The results of the study reveal: i) a good heat flux is achieved

(around 1 MW/m²); ii) the relatively uniform temperature distribution over the engine structure acts to reduce knock and increase power; iii) narrowing the cooling jacket clearance improves the heat transfer coefficient at low levels of heat flux, but the magnitude reaches a peak, while heat flux declines (where the maximum value was 150 kW/m²); iv) the core and fan size are both reduced by around 15%; v) at high loads, the boiling point (which influences detonation) is reduced by a reduction in the vapour pressure from increased heat transfer in the condenser and this also improves volumetric efficiency, increases torque, reduces friction and improves combustion; vii) conversely for low loads, the boiling point is increased by reducing heat transfer in the condenser and therefore increasing the vapour pressure; viii) fuel consumption is improved owing to the rise in coolant temperature, which is greatest at low load, but at high load this benefit is offset by worsening volumetric efficiency; and iv) a 10°C change in coolant temperature results in a 10°C change in liner-wall temperature, suggesting that very close control is possible.

A further study sponsored by Nissan [53] published in 1987, reported on the findings of an experimental study involving a novel EC system installed on a 206 kW inline 6 cylinder diesel engine. In this configuration, the condenser was controlled using a fan clutch with the steam temperature as a feedback signal. The coolant supply was controlled by an electrically-powered liquid pump with sensors to monitor the coolant level. Part of the study proposed a simple control method for an EC system for the engine of a concept truck. The conclusions are consistent with other previous studies namely that the benefits of EC systems are: i) a reduction in the size of heat exchanger and cooling fan; ii) no increase in the maximum combustion-chamber wall temperature; iii) a reduction in fuel consumption especially at part-load; and iv) a significant improvement in engine warm-up performance. A schematic diagram of the system proposed by Nissan is shown in Figure 8 where it can

be seen that the condenser is located at a high point in the system. This position was chosen to dispense with the need for a circulating pump, and instead, allows the condensed water to be returned to the cylinder block by gravity. System pressure would be controlled within limits using feedback to control the cooling fans.

Phase Three: Integration

In 1990 Lee et al. [54] published a comprehensive study of the development of an EC system for small four stroke engines. A test rig was built to compare the performance of an evaporatively-cooled engine with conventional liquid cooling. A schematic diagram of the test rig using a 0.667 L single cylinder engine is shown in Figure 9. The main findings of the paper (which confirmed the work of Leshner [6]) were that, compared with the liquid cooled equivalent, an EC system enhanced the brake power allowing for more rapid warming up, better fuel economy, and increased engine durability. The heat loss through the cylinder liner was also found to decrease with the use of an EC system.

Three patents were granted in 1992. The first [55] came from Volkswagen, and the concept is shown in Figure 10. This system concept used a liquid-vapour separator between the engine and condenser, and a liquid coolant by-pass to the condenser. It was claimed that the new architecture would allow sufficiently high engine temperatures so that vaporised coolant passes through the vapour line and separator into the main condenser. All the coolant passing through the condensate line also enters a heat exchanger used for vehicle heating. A further patent, granted to Toyota [56], proposed a new concept using a sealed cooling water recirculation system and a reservoir tank. The third patent granted for 1992 came from Eastman Kodak [57] bringing their experience of evaporative processes to bear on the problem.

Two more EC system patents were granted in 1992. The first by Sausner and Mertens [58] proposed adding a surge tank to prevent an engine from being affected by low

ambient temperatures. The second patent came from Volkswagen [59] and is shown in Figure 11. This used a tank, sealed from atmosphere, and partitioned by a diaphragm into an air-equalizing chamber and a coolant chamber. The system also included a low flow-resistance connection to the suction-side of a condensate pump. In 1993, Cummins, in collaboration with Georgia Institute of Technology [60], published the findings of analysis and experimental work on cylinder head cooling. This included detailed temperature measurements of a cooling jacket, and a finite element heat transfer analysis to identify regions of pure (single phase) convection, nucleate boiling, and film boiling. The cylinder head was divided into zones as shown in Figure 12a. Flow separation and stagnation points around the injector and valve sleeves of one cylinder are shown in Figure 12b. The conclusion of the study was that there was considerable uncertainty in the modes of heat transfer, particularly the boiling heat flux. Even at very high loads, the transition from pure single phase convection to nucleate boiling was uncertain. The work reported in [61] (1993) examined a nucleate boiling based engine cooling system in a climate controlled wind tunnel. Two types of completely-filled or partially-filled nucleate boiling systems shown in Figure 13 (as will be seen in the classification in Section 3, as examples of Class 1 and Class 2 type systems) were tested on a 75 kW four cylinder inline gasoline engine. The relative merits of completely-filled over partially-filled systems were demonstrated, and it was claimed that completely-filled systems are smaller, simpler, and cheaper, with better heat transfer characteristics.

Then in 1993, a radically different type of evaporative cooling system was patented [62] using spray evaporative cooling – further described in [63]. The intended application being transportation, avionics, and spacecraft where adverse gravity conditions prevail. The merit being very high heat flux being possible in a compact package. Another spray

evaporative cooling patent was published in 2003 [64] intended for use in cooling of electronics. No spray cooling systems have however yet been applied to IC engines.

Returning to 1997, VALEO had proposed a concept [65], known as the 'Newcool' system. To take into account the risk of 'vapour blanketing' in fully EC systems, it used a small electric water pump absorbing 30 - 60 W instead of a conventional engine driven water pump (absorbing 1 - 2 kW). Using a 1.9L VW Golf TDI engine, it was claimed that the proposed system cooled the engine in 95% of working conditions by forced liquid convection, and the remaining 5% would be satisfied by nucleate boiling heat transfer. The system also resulted in: more homogeneous cylinder head temperatures, lower fuel consumption (resulting from the lower power requirements of the water pump), the use of smaller diameter hoses (costing 40% less), and improved thermal comfort in the passenger compartment. However, the size of the expansion tank was recognised as a disadvantage of the system. In fact, the disadvantages of EC systems are discussed in [7] (1997), namely: i) coolant expansion; ii) higher operating temperatures which may result in metallurgical problems or affect engine performance, and iii) the risk of vapour blanketing with consequential deterioration in heat transfer rates. It was suggested that a minimum flow rate supplied by an electric pump be specified so that single phase convective cooling takes place most of the time, and that EC would be used only for more severe operating conditions. A VW diesel engine was studied experimentally and along with 3D thermo-hydraulic calculations, showed that the most important requirement in an EC system is to balance the flow rate of the coolant between cylinders. This could be achieved by geometric modification of the gasket.

The last EC patent in the 20th century [66] was granted in 1997 for a system to maintain engine lubricating oil at a desired temperature. Several publications in the first decade of the 21st century further improved the understanding of EC systems. In 2001, Kandlikar [67]

studied, for various refrigerants and water, flow boiling heat transfer in mini-channels i.e. with hydraulic diameter D_h in the range 0.1mm – 3mm. This work was undertaken to compare the values of heat transfer coefficient with correlations developed for conventional channels ($D_h > 3\text{mm}$). This was followed by two very substantial reviews of nominally conventional engine cooling technology. The first undertaken by Pang and Brace [1] in 2004, comprised a comprehensive review of engine cooling technologies for modern engines. They reviewed contemporary approaches to engine cooling, addressing such metrics as high and low temperature set points. They also examined ‘precision cooling’ and ‘split cooling’ systems, and concluded that the integration of split cooling (involving local coolant flow control to different parts of the engine) with precision cooling (the minimum cooling needed to achieved optimised temperature distribution) had the strongest potential to provide the desired level of engine protection, while gaining improvements in fuel economy and emissions. These precision cooling systems are considered to be substantially state-of-art ‘convectional’ systems. A different review was undertaken by Ap and Tarquis [2] in 2005 focusing on a number of particular trade-marked systems available spanning the state-of-art convectional cooling (but also as will become apparent, spanning Class 1 and Class 2 type systems within the classification discussed shortly in Section 4). The emphasis of this latter review being placed on four system attributes namely: relative weight, packaging and costs; fuel economy and emissions; thermal comfort; and the heat performance of each cooling system.

Three further EC system patents have been granted within the past decade, claiming in particular to achieve robust control. In 2005 for example, Siemens proposed a new method [68] for adjusting coolant temperature in which the coolant circuit contained an electrically driven pump and a controllable bypass valve. They claimed that when the coolant temperature set point changes abruptly, the coolant pump speed will rise during the short

interval in order to reduce the 'dead time' required for adjustment. To regulate the bypass valve, a controller was suggested to take account of the 'dead times'. In 2010, a patent [69] granted to Toyota, proposed a new method for robust control of engine output using at least two actuators. And a third patent, i.e. an application from General Electric [70] in 2014, focused on an integrated engine cooling system.

Two publications in 2015 have proposed the use of EC systems for automotive applications, although interestingly, neither for combustion engine cooling. One publication [71], examines the use of EC for a fuel cell stack intended for use as a range extender in a London taxi. This design avoids separate cooling channels allowing a change in the method of manufacture from the (currently expensive) etching process to pressing of metallic plates. The other publication [72] proposed EC for the hydraulic retarder in a heavy duty vehicle. Since this absorbs significant amounts of energy during the braking process, large amounts of heat need to be dissipated. Previous designs have integrated the retarder cooling into the engine cooling which has proved to be unsatisfactory. The proposed cooling system is specific to the retarder and was shown to be significantly more effective. Lin and Sunden [3] published a literature survey on vehicle cooling systems for reducing fuel consumption and CO₂ emissions. This included not only those for engine cooling but also air conditioning, cooling of electronics, and cooling of parts affected by friction. Their review collected together valuable results, and a discussion drawing on the work of more than 65 references.

Having summarised the literature, with EC systems as the focus, before identifying the research challenges, it is now appropriate to identify a number of distinct EC system concept classes.

4. A CLASSIFICATION OF EVAPORATIVE COOLING CONCEPTS

From the previous historical review of the EC literature plus some general consideration of energy and coolant thermodynamic properties, it is possible to identify four essentially different EC concept classes as follows:

Class 1: This comprises a sub-cooled system where the hot side volume is fully liquid flooded. The flow is driven around the system by a circulation pump. There are contributions from both single phase convection and boiling to the total heat transfer. However, there is a substantial increase in the contribution of boiling compared with existing sub-cooled systems currently in use for engine cooling. There are a number of concept differences compared with these existing sub-cooled systems: i) the deliberate provision for steam-venting from the engine to an expansion tank, ii) condensing wet steam in the bulk coolant and also the expansion tank, and iii) subsequent management of condensate by reintroduction to the cooling liquid, either before or after a heat exchanger. These systems can be characterised by various operating parameter values, such as a specific coolant volumetric flow rate of around 1.0 l/kWh, a coolant pressure in the region of 2.4 bar (absolute), a maximum coolant circulation power of 1% of the rated engine power, and a 7°C to 10°C temperature difference across the cooling system. These values will change significantly with reducing load.

Class 2: This is a full EC system in which heat transfer takes place only through a change of phase. Water is metered into the engine cooling circuit at saturation temperature and is allowed to completely boil off so the hot side volume is partially liquid and partially vapour flooded. Full provision is necessary for steam-venting from the engine. The concept has a heat exchanger which acts as a condenser. The flow is driven around the cooling circuit by a small circulation pump, and the flow rate is strictly regulated so that the hot side only receives the flow required for full boiling. The hot side coolant

specific volumetric flow rate is around 0.17 l/kWh at rated-power, and the maximum coolant circulation power is around 0.1% of the rated engine power with negligible change in the coolant temperature (owing to use of a condenser). For part-load conditions, it is expected that the hot side coolant specific volumetric flow rate will remain around 0.17 l/kWh, with coolant pressures being controlled to a maximum of 2.4 bar (absolute), and the coolant circulation power still around 0.1% of the engine power, or less.

Class 3: This is also a fully evaporative system concept in which engine cooling is achieved only by changing the dryness fraction of vapour passing through the cooling jacket. Unlike the Class 2 system, wet vapour is introduced into the cooling jacket. The heat transfer to the vapour increases the dryness fraction to just below 100%. The system hardware uses a heat exchanger, which does not condense the steam back to liquid, but to wet steam with the lowest acceptable dryness fraction. In principle, the effectiveness of a Class 3 system, stems from the similar magnitude of the latent heat in vaporising a mass of water droplets (in wet steam) as compared with the latent heat in vaporising the same mass of water (immersed in saturated liquid). The potential advantage of Class 3 is better flow control because it only has vapour flowing through all parts of the system. The difficulty of cooling hot surfaces by vaporising water droplets in wet steam, is to ensure sufficient droplets make contact with the hot surfaces to achieve the desired heat flux. This requires impinging jets of wet steam in the vicinity of the hot surfaces. As a consequence, significantly more power may be needed to drive the vapour through the system. Moreover the actual surface heat transfer rate is governed by the fundamental heat transfer process between the surface and adjacent fluids (e.g. conduction and convection but not the subsequent evaporative heat dissipation process which indirectly impacts on the heat transfer rate by changing the temperature gradient). Therefore, the

poor thermal properties of the wet gas compared to the saturated liquid will lead to a lower cooling rate performance compared to the Class 2 system.

Class 4: This is a system class based on exploiting a cooling concept involving spray evaporative boiling. Several concepts have been proposed for transport and spacecraft cooling problems [62-63] and for cooling electronic components [64]. In an adaptation of this system class, an atomised water cloud is sprayed in through nozzles into the engine cooling jacket. On making contact with the hot metal surfaces the atomised water completely evaporates. This steam is vented from the engine and passes through a condenser. The condensate is then pressurised by a pump and supplied to the cooling nozzles. The mass flow rate and circulating pump power is substantially less than for Class 1 (even though a high pressure is required to atomise the coolant).

Associated with this classification, it is appropriate to consider some form of evaluation and ranking of proposed designs. Table 2 shows a ranking in terms of cooling system class attributes, with a score of 5 being best, and 1 the worst.

Table 2. Cooling system class attributes

Cooling System	State-of-art Convection	EC Full Liquid Class 1	EC Part Liquid Class 2	EC Wet Vapour Class 3	EC Full Vapour Class 4
Attribute (5= best, 1=worst)					
Warm-up time	1	3	4	5	5
Pump Power	2	3	4	1	5
Sensitivity to vehicle inclination	5	5	1	4	5
Corrosion challenges	5	4	3	1	3
Cavitation	5	3	2	1	2
Response to transients	1	2	3	4	5
Noise, vibration and harshness (NVH)	5	2	1	3	4
"Failsafe"	5	4	2	3	4
Complexity	5	2	3	1	4
Cost	5	2	3	1	4
Total Score	39	30	26	24	41

Table 3 shows an assessment of the different cooling system hardware requirements. Both tables are constructed partly from prior (unpublished) work. The various attributes in Table 2 for example, such as warm-up-time, cost, etc. are largely adopted from state-of-the-art convective systems with some necessary assumptions for evaporative systems.

Table 3. Cooling systems hardware requirements

Cooling System	State-of-art Convection	EC Full Liquid	EC Part Liquid	EC Wet Vapour	EC Full Vapour
Hardware		Class 1	Class 2	Class 3	Class 4
Coolant Jacket(s)	Split	Split	Split	Split	Single
Pump 1	✓	✓	✓	✓	✓
Pump 2	X	✓	✓	✓	✓
Hotside Spray	X	X	X	X	✓
Thermostat(s)	1 or 2	1	1	1	X
Radiator & Fan	✓	✓	✓	✓	X
Condenser & Fan	X	X	✓	✓	✓
Condensate return(s)	X	X	✓	✓	✓
Anti-surge systems	X	X	✓	X	X
De-aeration system	X	✓	✓	✓	✓
Filler cap (or sealed)	✓	Sealed	Sealed	Sealed	Sealed
Expansion tank	✓	✓	✓	✓	✓
Cab Heater Circuit	✓	Liquid	Liquid	Liquid	Vapour
Transmission oil Cooler	✓	Liquid	Liquid	Liquid	Vapour
Temperature Sensor(s)	1 or 2	2	2	2	2
Coolant level sensor(s)	1	1	2	1	2

Classes 2, 3, and 4 are likely to be of most use around the cylinder head zone where there are relatively high levels of heat flux (in the region of 1 MW/m^2). Below this level of heat flux, a combined (single phase with EC) system may offer a more practical solution. The relevant design questions concern the total system mass, the particular coolant mass flow rate, the power required for coolant circulation, the heat rejection from the condenser or cooler, corrosion resistance, and the overall controllability of the system under dynamic conditions. Further work is also required to understand the influence of surface

topography, wetting systems, and possible benefits of directing coolant, rather than allowing it to flow in a more general sense.

Regarding the examples of the different classes arising in the literature review of Section 3, particular designs would probably not have been considered by most of the authors to belong to any of the identified Classes 1 - 4 of Table 2. Moreover there was probably a deliberate (and understandable) intention not to reveal the exact system details in order to protect commercial intellectual property rights. However from the literature review the system described in Figure 4 (i.e. fitted to the cooling hopper) is an embodiment of an open evaporative system with a recuperating condenser, and can therefore be considered to belong to the Class 1. The Leshner system [6] by contrast can be considered to belong to Class 2 as the associated patents show partially filled coolant jackets. This is because the coolant jacket remains substantially filled to ensure adequate coolant for local evaporation. Similarly the system described by Figure 5 belongs to Class 2 in that the cylinder head coolant jacket has expansion volumes and vents. The 1987 concept produced by Nissan [49] and [53] includes both Class 1 and Class 2 systems. By contrast, the systems described in [55][58] and [59] only belong to Class 2. No proposed systems appear to belong to Class 3 whereas two systems described respectively in [62][63] and [64] belong to Class 4.

With the history of EC literature reviewed, an EC system classification in place, and various system attributes established, a position has been reached where the remaining research challenges can be identified. These are the challenges that need to be addressed in order to remove the obstacles to development of EC systems by vehicle manufacturers.

5. THE RESEARCH CHALLENGES FOR EVAPORATIVE COOLING SYSTEMS

In addressing high level research requirements for EC systems, it is appropriate to return to that critical question: why have the proposed and apparently functional EC systems not been developed to the extent that they are now used in road vehicles? It is for example, particularly surprising that the system reported in [52] never made it to production. The answer to why none of the systems discussed were taken up, point to several significant challenges yet to be addressed. In identifying these major challenges, the focus can actually be pitched at two levels: i) at the specific system design level for the different classes identified, including questions such as the practicalities of thermal management, stability of control, and the sensitivity of system performance to real operating conditions; and ii) at the level of the fundamental physics, in particular to the current understanding and general predictive capabilities associated with the heat transfer processes and mechanisms in systems with two-phase flow. Starting with the EC system design issues associated with the different classes, the general questions concern:

- i) Which system offers the best overall heat transfer?
- ii) Which system is best from a controllability viewpoint when disturbed by real vehicle dynamics, engine noise, and vibration?
- iii) Which system is best in terms of simplicity, robustness, and durability?
- iv) Which system provides the best distribution of energy? and
- v) Which system offers the most cost effective solution?

To answer these questions a series of related challenges must be overcome. One of the biggest challenges is to achieve the high levels of flux (of around 1.25 MW/m^2) with metal temperatures up to 240°C maximum for aluminium alloy coolant jackets. A second major issue is how to manage the generated steam within the engine cooling jacket, which is particularly relevant to Class 1 systems. This does not make for an easy transition to EC systems from existing (largely convective) cooling systems. Also, of particular concern for

Class 1 systems, is the uncertainty of the vapour ‘pumping’ mechanism (i.e. whether it is forced flow, or the result of buoyancy variations), and second, the role of mechanically induced agitation. To size a system to ensure the benefits generate a sufficiently advantageous margin (particularly achieving improved fuel economy), it is necessary to go through an iterative design process of modelling and refinement. The study in [7] found that the most important requirement for an EC system was to balance the flow rate of the coolant between cylinders. The intended flow distribution is very likely to be significantly influenced by agitation from engine vibration and vehicle motions. Elsewhere, in [65], the size of the expansion tank was found to be a serious disadvantage. Therefore careful system design will be essential if the benefits of EC are to be realised. To summarise, the main challenge for Class 1 systems will be stable management of boil-off, since the task of heat dissipation is split between two different heat exchangers. For Class 2 systems, because heat is rejected through boil-off only, the main challenge will be efficient peak hot-side metal temperature management through control of rapid (relatively large amplitude) variations in vapour pressure within the coolant jacket. For Class 3 systems, the main challenge will be design of a vapour heat exchange system to achieve the required heat transfer associated with wet steam vapour. For Class 4 systems, which unlike the other three classes, involves spray evaporative cooling, the main research challenge will be to accurately predicting heat transfer coefficients, multivariable control stability, and subsequent system optimisation.

Turning to the challenges associated with current understanding of the fundamental physics, the published literature clearly suggests that good predictive capability is required to refine any design before committing to hardware. It also points to a lack of certainty in quantifying the heat transfer coefficients and boiling regime boundaries at engine related conditions and passage geometries. Therefore the most likely reason for the lack of EC

systems in production today would appear to stem from this lack of acceptable predictive capability. This assertion is supported by the analytical and experimental study in [60] which points to considerable uncertainty in the values of the heat transfer coefficients. Bringing all these factors together the research challenges in creating a robust system can be listed as follows, namely that there is a need:

- i) To set a detailed design specification for EC systems in terms of operating temperatures, pressures, and heat transfer requirements.
- ii) To rigorously establish which of the system Classes 1 to 4 most acutely meets the required specification.
- iii) For an EC system design to fully meet the specification with cost effective control sensing, control actuation, power, and weight.
- iv) To better understand the physics of vapour formation in the presence of vibration and agitating boundaries typical of the levels and frequencies arising in a boosted combustion engine used for both on-road and off-road vehicles.
- v) For an experimentally verified heat transfer predictive capability for vapour formation in the presence of vibration and agitated boundaries.
- vi) To establish the control variables that will achieve peak heat flux levels comparable with state-of-the-art sub-cooled systems.
- vii) To understand the EC control problem in order to adopt a robust control strategy that will give absolute assurances of stability under all operating conditions.
- viii) To identify a durable coolant that does not suffer from deterioration as a result of repeated evaporation and condensing.
- ix) To maintain anti-corrosion and anti-erosion cooling system standards.

6. CONCLUSIONS

Evaporative cooling systems proposed over the past century have been examined. Initially, the physical principles and benefits of EC have been expounded along with the state-of-the-art capability for the modelling and numerical simulation. Several proposed designs dating back to 1922 have been examined. Four EC system classes have been defined to help put historical concepts into context. A systematic examination of the critical

areas in EC systems has successfully identified the remaining research challenges for EC systems in automotive engine cooling applications. These remaining research challenges include: a reduction of the considerable uncertainty in the modes of heat transfer; gaining a better understanding of the physics of vapour formation in the presence of vibration and agitating boundaries typical of the levels and frequencies arising in a boosted combustion engine powering on- and off-road vehicles; and for a more fundamental experimentally-verified heat transfer and heat flux predictive capability for vapour formation in the presence of vibration and agitated boundaries. If these research challenges are overcome it will then be possible to establish the control variables that will achieve peak heat flux levels comparable with state-of-the-art sub-cooled systems. Finally it will allow the EC control problem to be understood in order to adopt a robust control strategy that will give absolute assurances of stability under all operating conditions.

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FIGURES

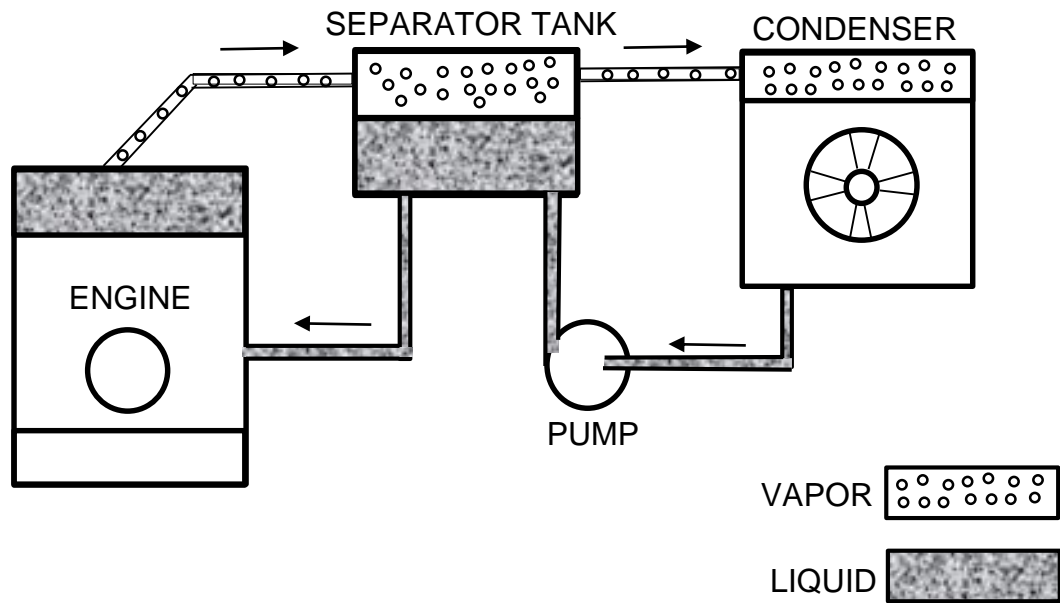


Figure 1. Schematic diagram of an automotive evaporative cooling system.

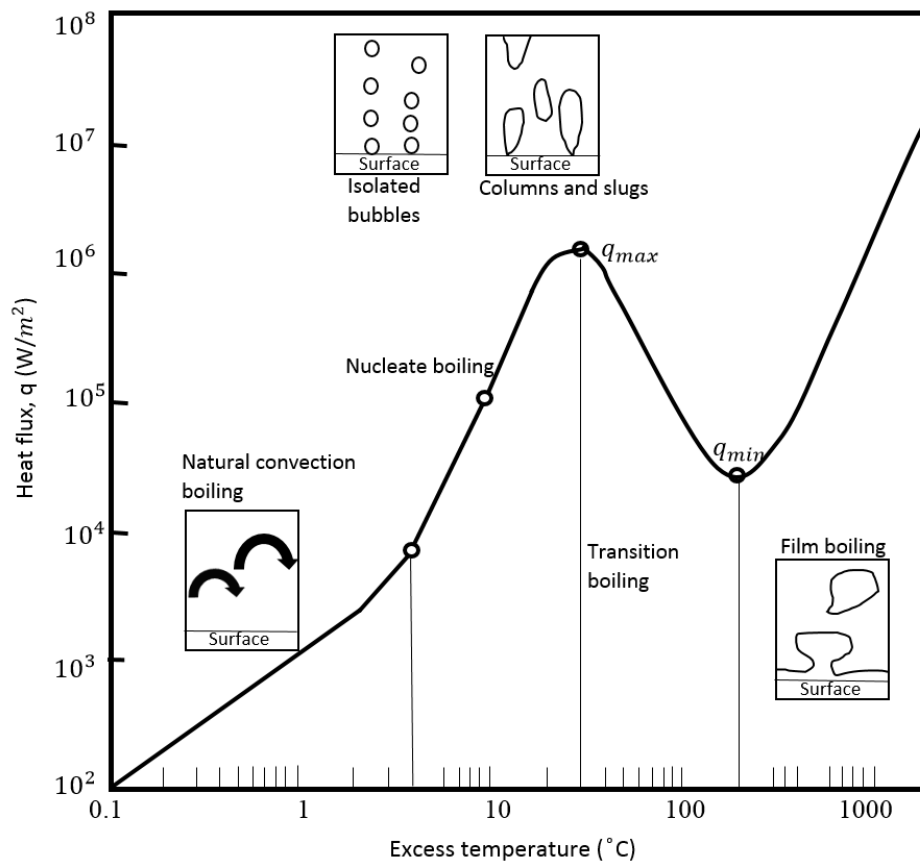


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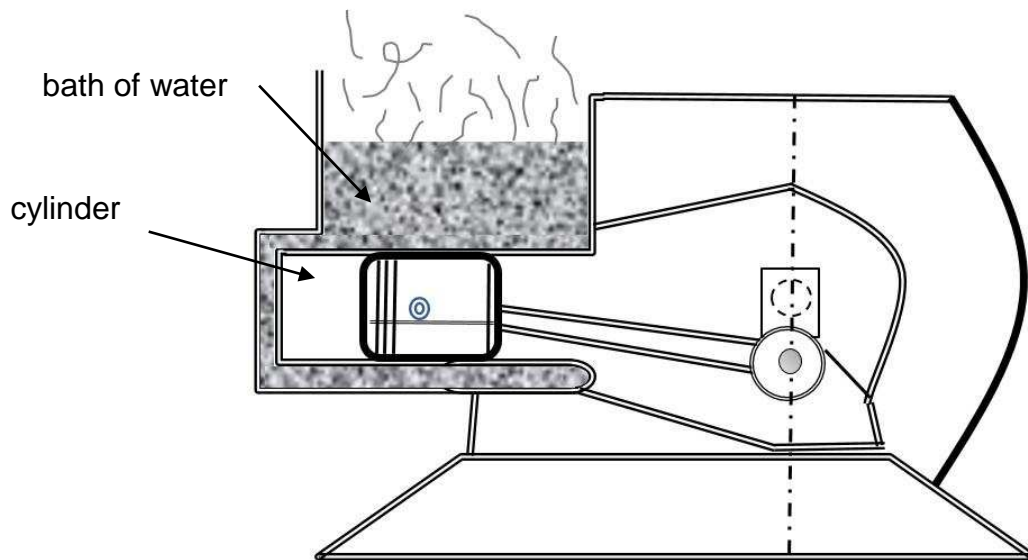


Figure 3. Evaporative cooling system in which the engine cylinder block is surrounded by a bath of water.

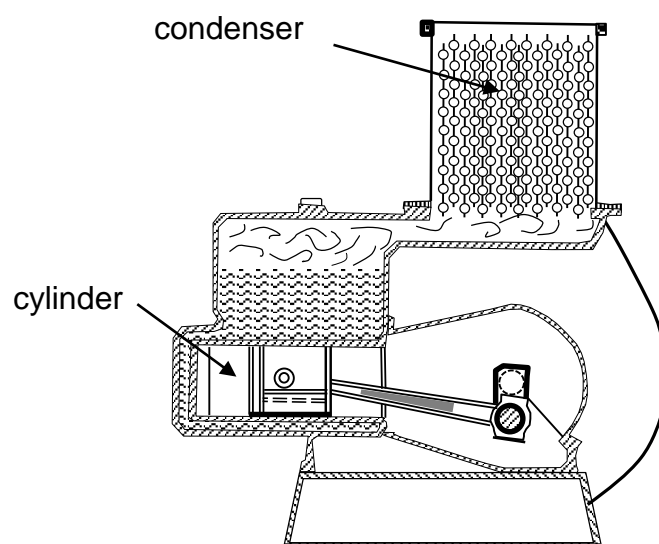
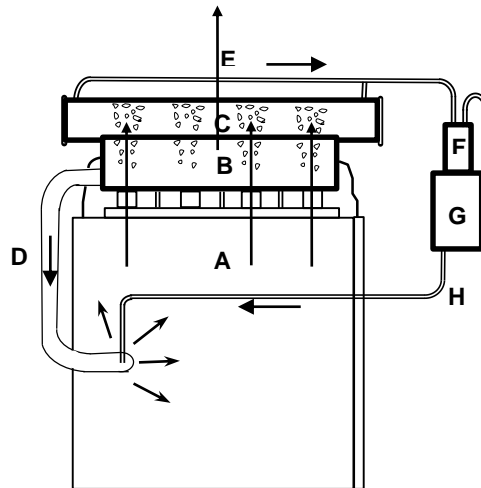


Figure 4. Evaporative cooling system using a steam condenser fitted to a cooling hopper.



- A. Engine block
- B. Steam separator
- C. Condenser
- D. Coolant return to engine block
- E. Condenser vent line
- F. Vent tank
- G. Filler and make-up tank
- H. Make-up line to engine

Figure 5. Schematic diagram of dual-circuit cooling [45].

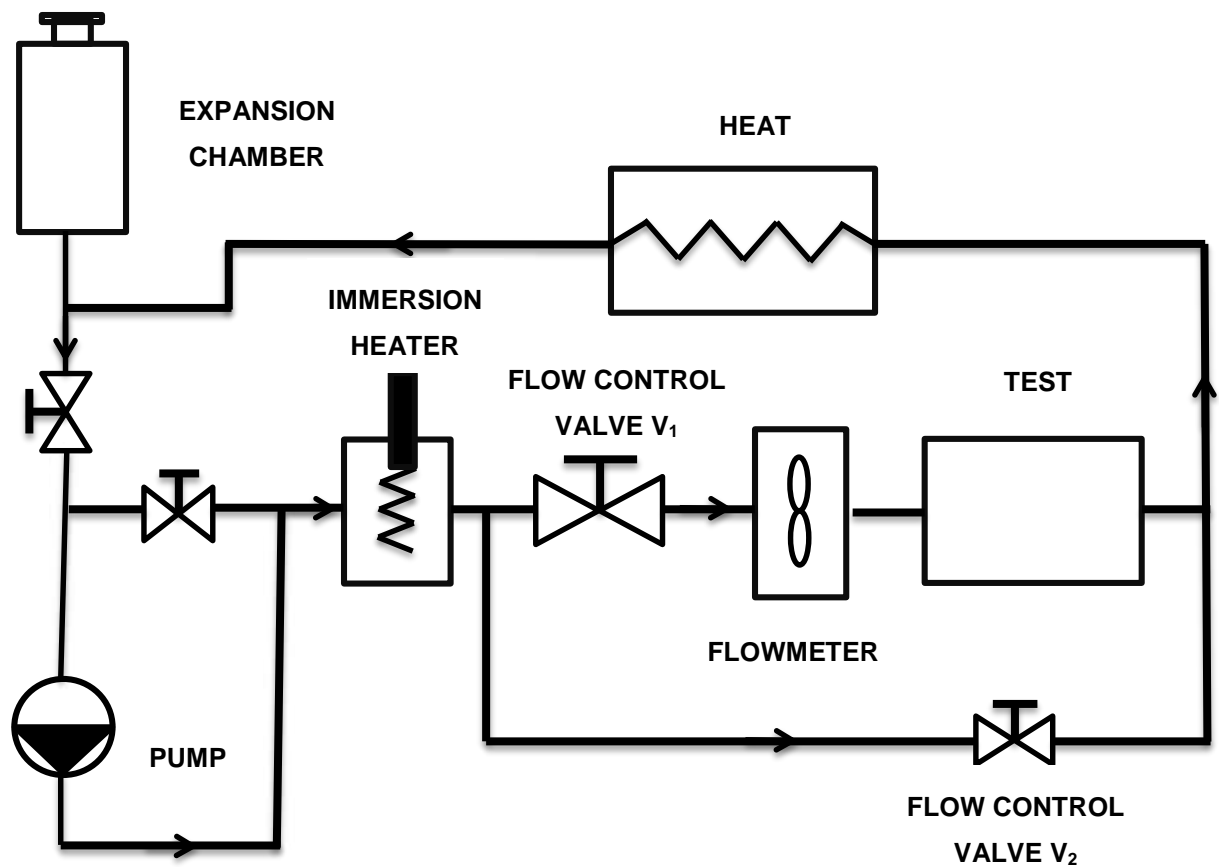
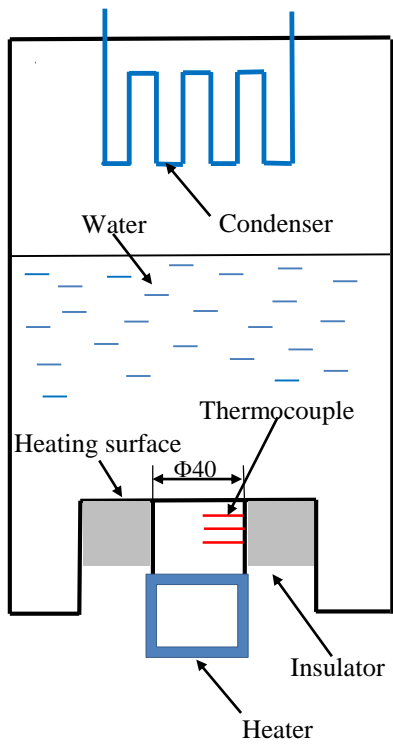
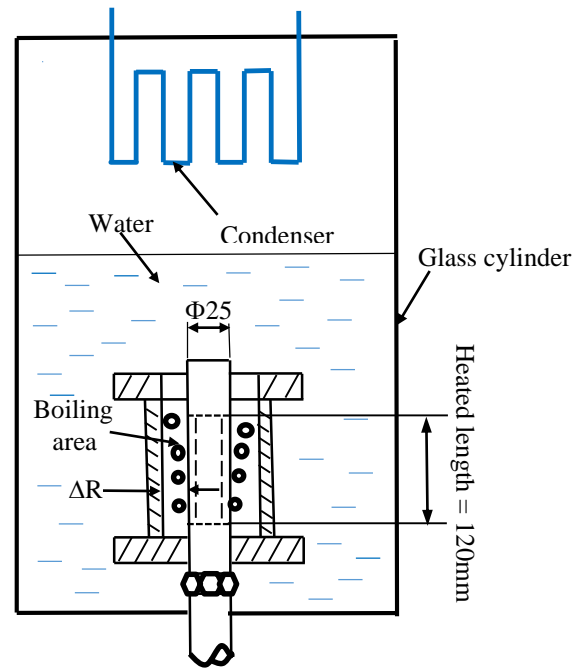


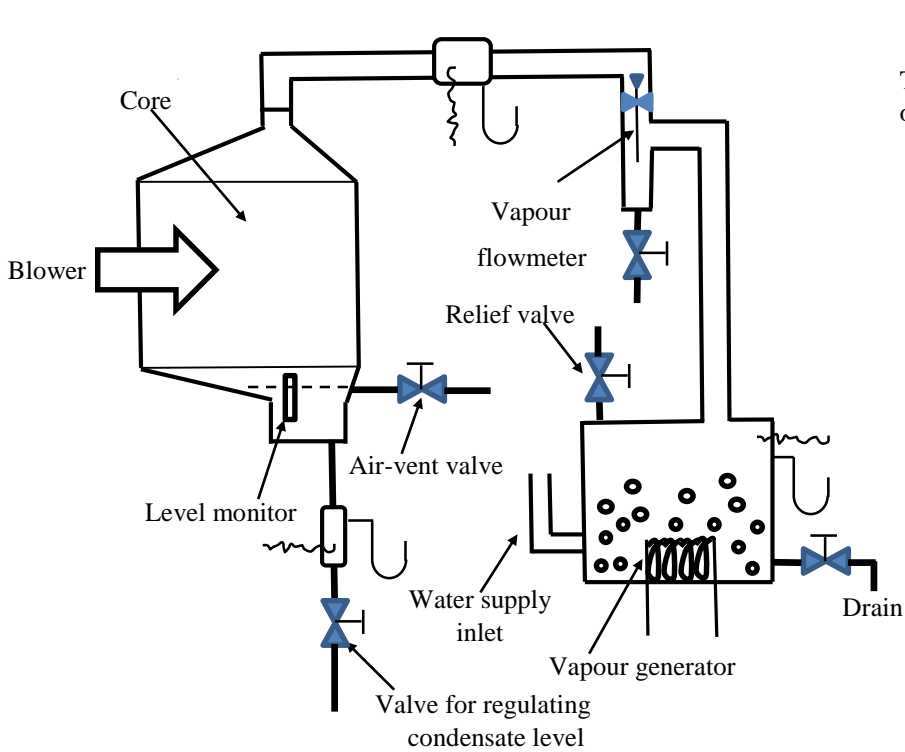
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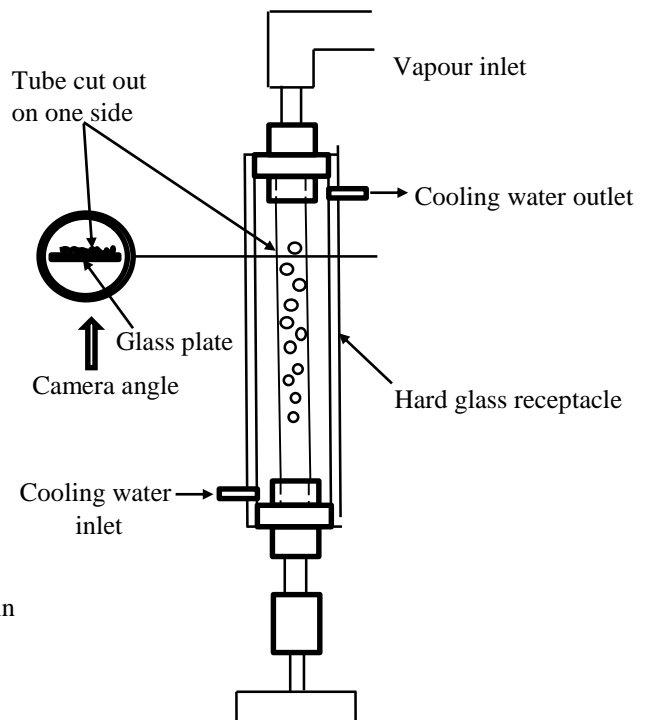
(a)



(b)



(c)



(d)

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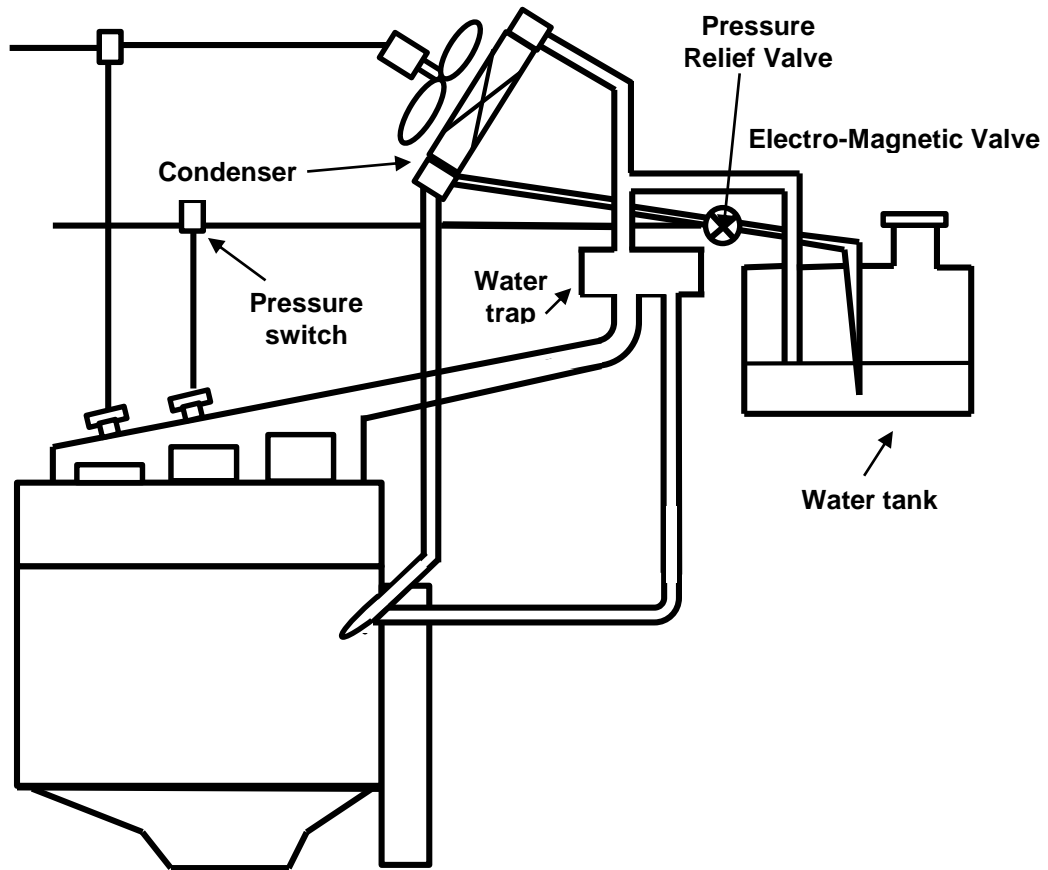


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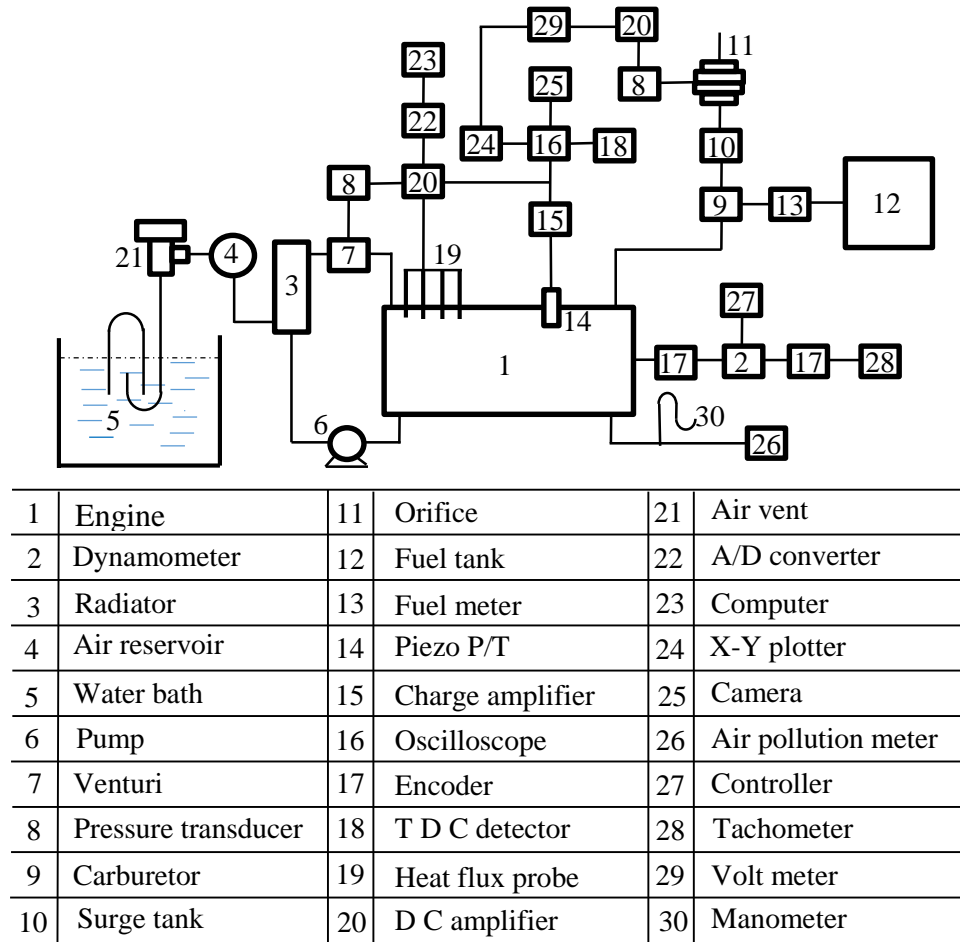


Figure 9. Schematic diagram of the experimental apparatus [54].

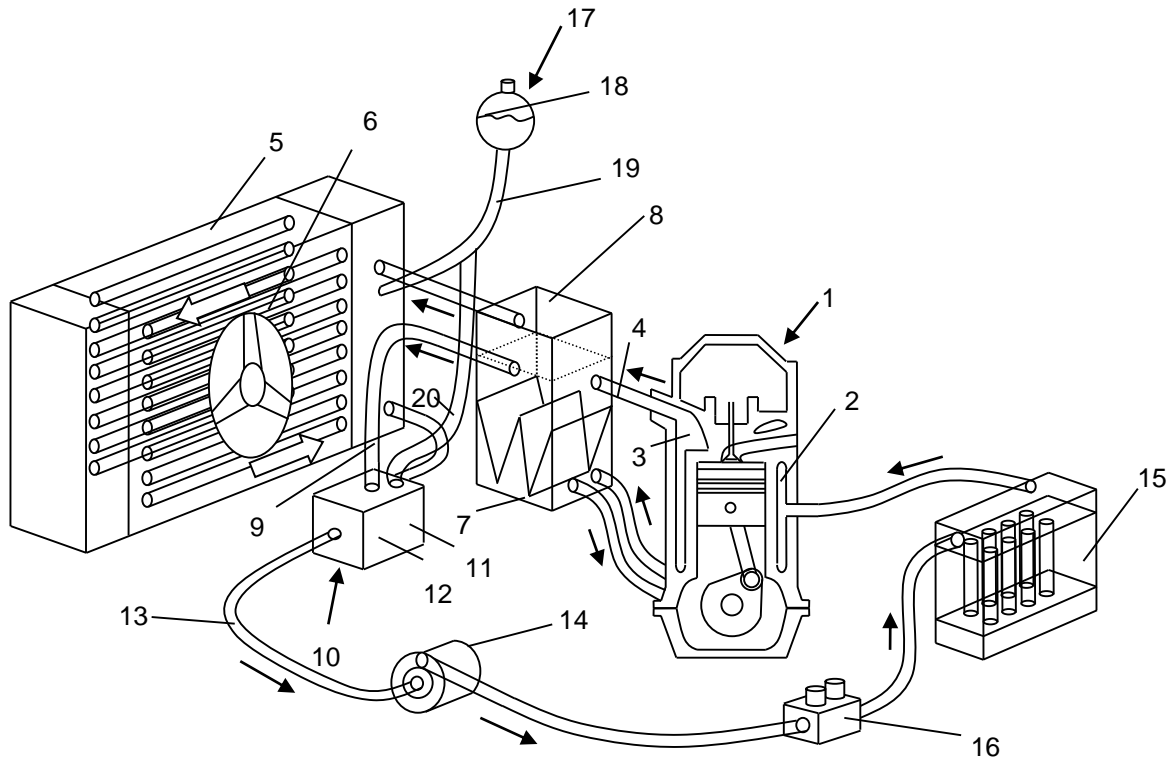


Figure 10. Schematic diagram of evaporative cooling system proposed by Volkswagen for which the labelling of the arrowed-markers can be found in [55].

Figure 10 components list	
Number	Name
1	Internal combustion engine
2,3	Cooling chambers
4	Vapour line
5	Main condenser
6	Fan
7	Evaporation oil cooler
8	Liquid-vapour separator
9	Bypass line
10	Mixer tank
11	Liquid level
12	Internal baffle
13	Condensate line
14	Condensate pump
15	Vehicle heater heat exchanger
16	Transmission oil cooler
17	Compensation tank
18	Diaphragm
19,20	Vent lines

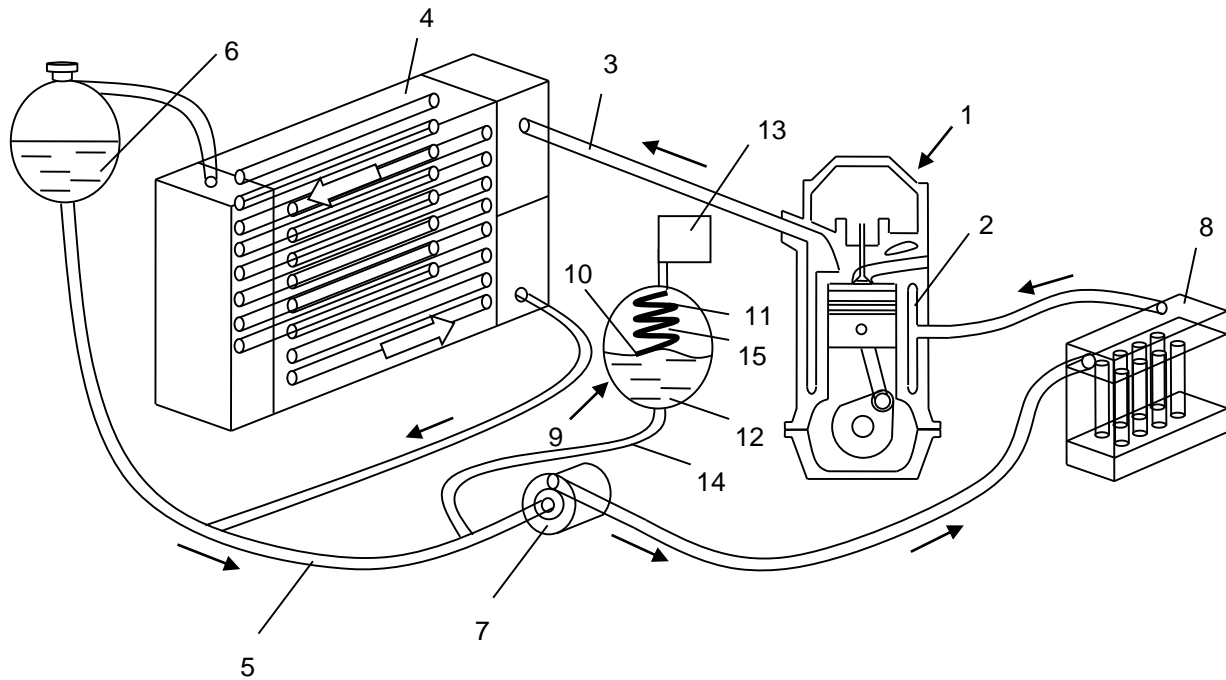


Figure 11. Schematic diagram of evaporative cooling system for an internal combustion engine having a coolant equalizing tank for which the labelling of the arrowed-markers can be found in [59].

Figure 11 components list	
Number	Name
1	Internal combustion engine
2	Cooling chambers
3	Vapour line
4	Condenser
5	Condensate line
6	Coolant supply tank
7	Condensate pump
8	Vehicle heating system
9	Coolant equalising tank
10	Membrane
11	Air chamber
12	Coolant chamber
13	Additional air chamber
14	Large flow cross-section line
15	Mechanical spring

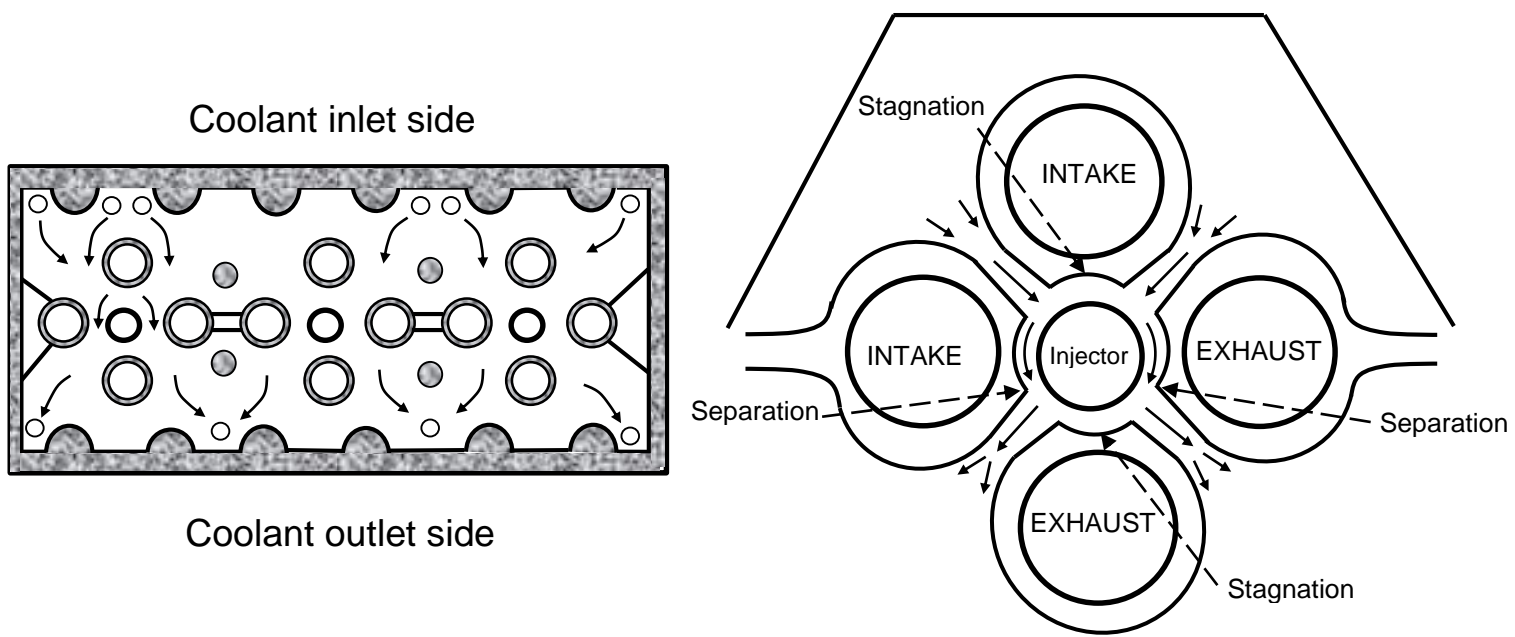


Figure 12. Schematic diagram of different cylinder head zones [60].

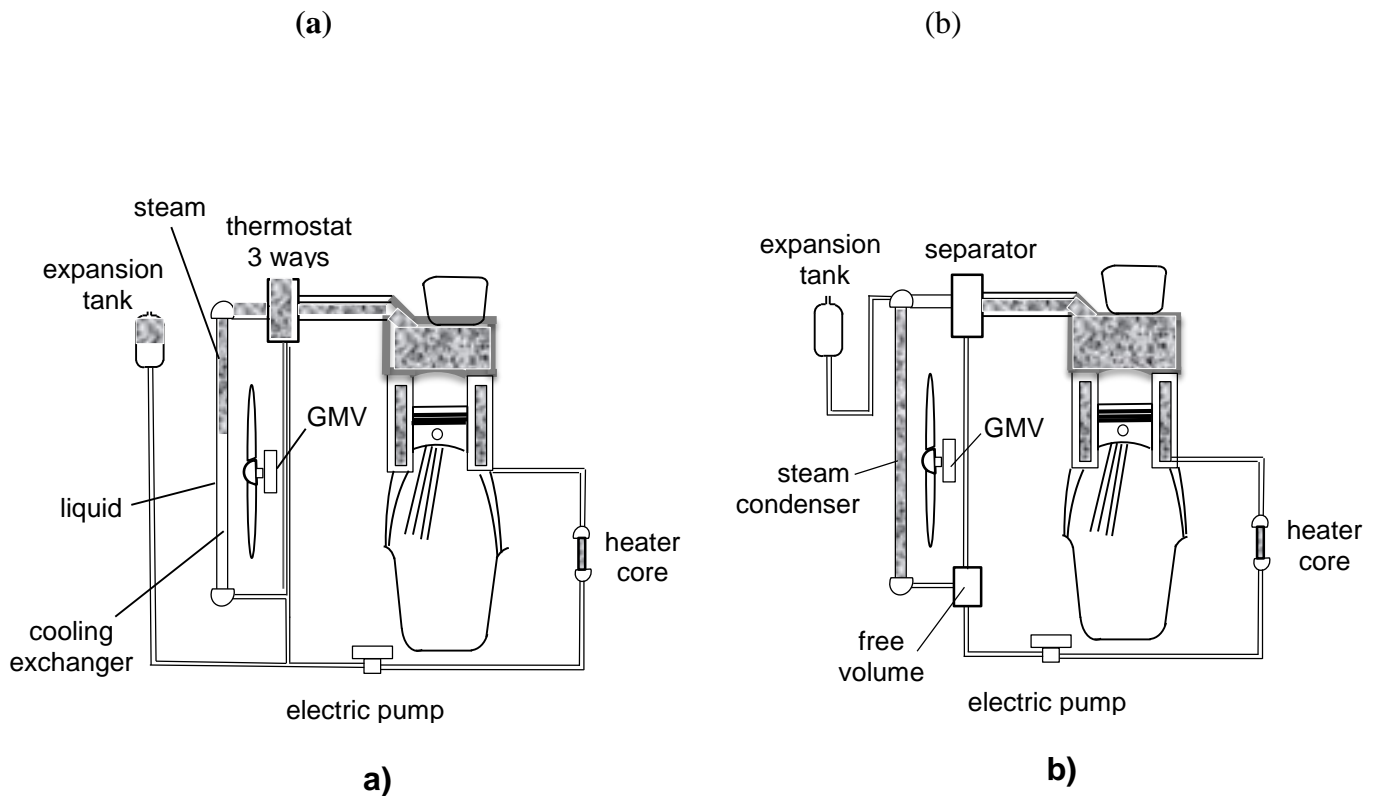


Figure 13. (a) Completely filled nucleate boiling cooling system, (b) partially filled nucleate boiling cooling system [61].